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Evaluation of machine tools with position-dependent milling stability based on Kriging model

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ABSTRACT

Machine tool's milling stability is a function of the frequency response function (FRF) at the tool tip, which varies with the position changes of the moving components within the machine tool work volume. The approach to obtain stability lobe diagram avoiding chatter vibrations is generally based on some specific positions, resulting in an incomprehensive and inaccurate chatter prediction. This paper presents a new method to rapidly evaluate the position-dependent milling stability of machine tools. In this method, position combinations of the moving components are arranged to perform the impact testing, and then the corresponding identified FRFs at the tip of machine tool frame-holder base are acquired to identify the modal parameters, with which a Kriging model is developed to describe the relationship between positions and modal parameters. Consequently, based on the modal fitting technique, tip FRFs of machine tool frame-holder base at any position in the whole working space can be reorganized. And such tip FRFs are further adopted to calculate the tool tip FRFs with the identified contact parameters of the holder-tool system using the receptance coupling substructure analysis (RCSA) technique. Then, the milling stability at different positions can be assessed. Furthermore, the proposed method is applied to a vertical machining center, and its accuracy for predicting the milling stability in entire work volume is validated through chatter tests.

1. Introduction

Chatter occurring in machining process is a major problem that causes poor surface finish, low productivity, increased tool wear, machine tool failure, serious noise and increased cost in machining application [1–3]. As a consequence, machining stability has been researched into deeply for decades, and generally the developed stability lobe diagram is utilized to select an appropriate combination of the depth of cut and spindle speed [4–7]. In order to obtain stability diagrams, FRFs at the tool tip should be determined initially. Recently, many researchers are focusing on the spindle dynamics, including dynamic modeling of the high-speed spindle system to obtain the speed-dependent tool tip FRF [8–10]. Besides the dynamic behavior of the spindle system, during the machining operations, varying position combinations of machine tool's moving components give rise to the changing mass matrix, stiffness matrix and damp matrix of whole machine tool, which may leads to the uncertain tool point FRFs as well [11–13]. Therefore, utilizing the tool point FRFs achieved at one specific position to depict the stability diagram to represent the machining stability in entire working space may result in incorrect chatter predictions. Accordingly, the analysis of the machining stability in machine work volume is highly significant.

Recently, more and more researchers begin to research the machining stability of machine tools based on multi-position and two types of methods are used in general. One method is based on experiments. In the study of Yang et al. [14], the tool point FRFs with different feed directions and spindle positions were obtained based on the impact tests to investigate the stability lobe diagrams, which were combined with the chatter tests to analyze the chatter reduction and cutting parameters selection. However, it is costly and time consuming to repeat the impact testing at tool tip for every position and holder/tool change during the application [15]. To minimize experimentations, another method based on finite element (FE) models of machine tools is implemented in evaluating the position-dependent response.

Since FE models have very large order, response analyses of whole machine tools are also computational costly. In addition to the varying position, it requires cumbersome re-meshing strategies to meet the boundary condition at any discrete position, which is time consuming in practice as well [16,17]. The developed co-simulation techniques with FE

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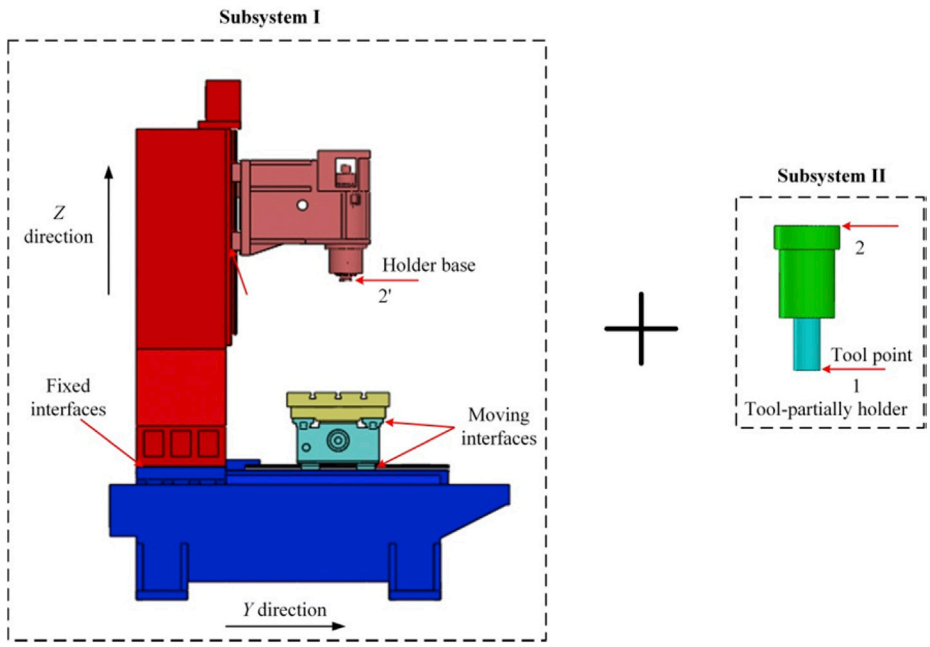

Fig. 1. Two subsystems of the machine tool. Tool-holder response (subsystem I) is coupled to the position-dependent response of machine tool frame (subsystem II).

solvers coupled to multi body simulation codes can reduce time, whereas they are restricted to flexible bodies attached at geometrically fixed contact points [18]. Consequently, researchers devote themselves to analyze the position-dependent machining stability for multiple flexible bodies [19,20]. One of the most typical researches is presented by Law et al. [20], in which a reduced model substructural synthesis approach is developed to model the entire machine tool system, allowing a rapid investigation on the relationship between positions and machining stability. However, the connections between the tool-tool holder-spindle and between the spindle-spindle housing are assumed to be rigid. Huang et al. [21] and Zhang et al. [22] combined response surface method, experiment design method and FE analysis to establish a mathematic model for predicting the minimum critical axial cutting depth at any position. Since this approach is time prohibitive, it cannot be used to obtain stability lobe diagrams and the mathematical model should be reestablished as the holder-tool system changing.

This paper presents a new method to facilitate efficient prediction of tool tip FRFs within machine tool work volume, avoiding complicated theoretical calculation and impact modal tests at each tool change. First, a mathematical model for predicting the modal parameters of tip FRFs of the machine tool frame-holder base at any position is established based on the Kriging theory and the arranged impact testing. Then, these predicated modal parameters are utilized to reorganize the corresponding FRFs based on modal fitting technique. Thus, with the identified connect parameters of holder-tool system, tool tip FRFs are calculated through receptance coupling substructure analysis and further used to calculate stability lobe diagrams to investigate the position dependent machining stability.

Henceforth, the remainder of this paper is organized as follows. The principles of the proposed method for predicting position-dependent machining stability is presented in Section 2. In section 3, applying the proposed method to a three-axis vertical machining center, the stability diagrams are examined through some case studies. Finally, conclusions from the current research are provided in Section 4.

2. Theoretical background

To efficiently achieve the position-dependent tool tip FRFs, a whole machine tool is divided into two subsystems as shown in Fig. 1. One is the whole machine tool frame (subsystem I) including the major components, namely: bed, column, worktable, saddle, headstock, spindle and holder base. The other subsystem is the tool-partial holder assembly (subsystem II). The tool tip responses can be calculated by coupling the end points responses of subsystem I to the responses obtained at the tip of subsystem II utilizing a receptance coupling approach, which allows obtaining dynamic responses of different tool-holder combinations independently and without having to repeat experiments or FE analyses on whole machine tool. Accordingly, the position-dependent tip FRFs of subsystem I should be determined initially, which can be identified by combining the Kriging method with modal fitting technique.

2.1. Modal fitting

According to the modal theory, the FRF is a superposition of individual modes, which can be determined using modal parameters as follows [23]:

$$H(\omega) = \sum_{r=1}^{N} \frac{1}{K_{er}\left[\left(1-\lambda^2\right) + j2\xi_r\lambda\right]} \tag{1}$$

where

$$\lambda = \omega/\omega_r \qquad \xi_r = \frac{C_r}{2M_r\omega_r} \tag{2}$$

If these individual modes are not closely spaced, when the vibration frequency ω equals the r_{th} natural frequency ω_r, the real and imaginary parts of the FRF are provided in Eqs. (3) and (4).

$$\mathrm{Re}[H(\omega)]_{\omega=\omega_r} = \frac{1}{K_{er}}\left(\frac{1-\lambda^2}{\left(1-\lambda^2\right)^2 + (2\xi_r\lambda)^2}\right) \tag{3}$$

$$\mathrm{Im}[H(\omega)]_{\omega=\omega_r} = \frac{1}{K_{er}}\left(\frac{-j2\xi_r\lambda}{\left(1-\lambda^2\right)^2 + (2\xi_r\lambda)^2}\right) \tag{4}$$

Neglecting ξ_r^2, the maximum and minimum value of the real part approximately will occur at the vibration frequency $\omega_1^r = \omega_r\,(1\text{-}\xi_r)$ and $\omega_2^r = \omega_r\,(1+\xi_r)$ respectively. The imaginary minimum is at the resonant

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